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Development of a High Ambient Temperature Cooling Unit Based on Microcompressor Technology

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Development of a Cooling Unit for High Ambient Temperatures Based on Microcompressor Technology

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ABSTRACT

A good matching between size and efficiency is a desirable design feature of cooling units for electronic systems, due to high capacity to size ratio and small weight. For this purpose, Embraco developed a new refrigerating system for small enclosures and telecommunication stations, which makes use of a new miniaturized linear microcompressor. The air-conditioning cooling unit consists of finned-tube heat exchangers and a direct current (DC) oil-free microcompressor that uses R-134a as the refrigerant. The paper presents a detailed description of the cooling unit and performance tests. A novel refrigerating cycle for thermal management of the microcompressor was created especially for high ambient temperatures. In order to run all experimental tests, a cooling unit calorimeter was assembled. In the experimental facility, the ambient temperature was varied (25, 35 and 55 °C) and the bulk temperature inside the enclosure was controlled via an electric heater. Commercially available thermoelectric systems used for electronics cooling were also compared with the vapor compression system. The results have shown that the microcompressor unit presented a coefficient of performance (COP) approximately two times larger than thermoelectric solutions.

1. INTRODUCTION

The trend of miniaturization combined with high processing capabilities has increased the heat dissipation of electronic components, especially those with a high power input. The observed temperature increase leads to a drastic reduction of the lifetime of critical electronic equipment, such as those used for power distribution and communication systems. Additionally, overheating leads to a reduction of efficiency due to switching losses, which become relevant when operating with high electric voltages and currents.

In order to extract the heat dissipated by electronics, several types of air-conditioning refrigerating systems are found commercially for telecom stations and command panels. Different solutions are used for electronics cooling. However, most of the small units with low cooling capacities still employ thermoelectric systems due to size restrictions imposed by the application. Small size refrigeration systems are usually associated with lower thermodynamic efficiencies. However, according to Barbosa *et al.* (2012), compact high-efficiency vapor compression systems can make good candidates for electronics cooling due to high capacity by size ratio and light weight.

As pointed out by Marcinichen *et al.* (2010), the fact that some power distributors dissipate heat on the order of 5-15 MW makes the efficiency of the refrigeration system for electronics cooling an enormous environmental issue. In this context, the use of a compact vapor compression refrigeration system that uses a miniature-scale compressor is very appealing because of their features for higher performance and efficiency when compared to other solutions, such as thermoelectric coolers.

Several studies have been conducted with a focus on energy saving for industrial enclosures and telecom equipment (Nakao *et al.*, 1988, Rabie and Delport, 2001 and Choi *et al*, 2007). However, most of these studies are related with cooling systems that presented heat loads with the order of magnitude of kilowatts. To the present author's best knowledge, there are no studies available in the open literature for compact telecom cooling units as efficient substitutes for the thermoelectric solution.

Furthermore, telecommunications equipment in the desert can reach very high ambient temperatures, up to 55°C. Such extreme condition combined with the high capacity per size ratio results in an insufficient heat transfer area to keep the microcompressor below the shell temperature limit of 80°C. Above the temperature limit, the microcompressor loses its performance. Hence, a novel refrigerating circuit was developed in order to allow the microcompressor to operate under such extreme conditions.

This work describes a compact cooling unit to be used in electronics cooling under high ambient temperatures that uses an oil-free microcompressor as main component. Experimental tests were carried out in order to evaluate the cooling capacity and the coefficient of performance of the microcompressor unit. Furthermore, a performance comparison between the miniature-scale vapor compression unit and thermoelectric coolers for telecom stations was performed.

2. COMPACT VAPOR COMPRESSION COOLING UNIT

Among all main dimensions, the width of an outdoor mounted air-conditioning is the most critical due to side-by-side arrangements usually adopted. Thus, the unit shape was designed with the intention to keep the same width found in thermoelectric units.

Embraco cooling unit resulted in overall dimensions of 660 x 200 x 120 mm (height x width x length), with a total weight of 8 kg. The microcompressor cooling unit in its final shape can be seen in Figure 1.

Additionally, the chosen structural material that surrounds all components is acrylic and the insulating material that separates the evaporator side from the condenser side is polyurethane foam. Since heat infiltration through the unit is an issue that must be avoided, the thermal conductivity of the structural parts must be low.

The working fluid used for this prototype was R-134a, which is usually employed in most refrigeration systems for electronics because of its non-flammability. The capillary tube parameters and the refrigerant mass were obtained by means of evaluation tests to be further described. Standard 1/4" tubing was used to connect the refrigerating components due to its high mechanical flexibility inside space gaps found during the assembly.

The microcompressor unit was compared with two commercially available thermoelectric coolers, named as thermoelectric unit 1 and 2, respectively.

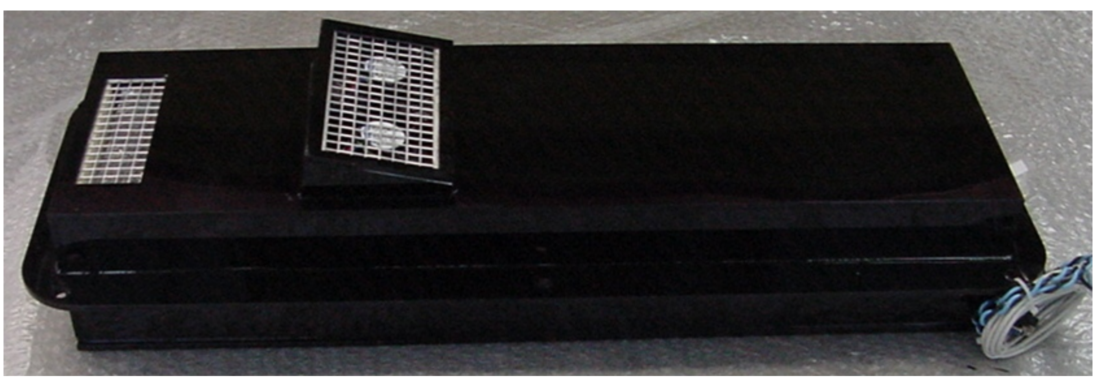

Figure 1. Microcompressor cooling unit.

2.1 Refrigerating Loop

The schematic illustration of the refrigerating circuit is shown in Figure 2. Besides the microcompressor and capillary tube, the cooling unit consists of four heat exchangers. The so-called pre-condenser and post-evaporator are additional heat exchangers, which work together with the traditional evaporator and condenser.

The refrigerant pumped by the compressor as superheated gas flows to the pre-condenser, where the refrigerant partially condensates, leaving the heat exchanger as two-phase flow. Then, the fluid flows to the post-evaporator, which is a wrapped tube around the microcompressor shell. With the physical contact, the pos-evaporator extracts heat from the microcompressor in the form of latent heat, evaporating the fluid. After that, the refrigerant as superheated vapor flows to the condenser and the one-stage refrigerating cycle occurs normally. The pressure-enthalpy diagram of the proposed refrigerating cycle can be seen in Figure 3.

As mentioned before, the pos-evaporator comprises a serpentine tube in contact with the microcompressor, and the pre-condenser comprises a finned-tube heat exchanger, as evaporator and condenser. The condenser and the pre-condenser are part of the same finned device with independent circuits. Thus, both components share the same air flow stream.

The latent heat exchanged along the post-evaporator maintains the microcompressor shell temperature close to the condensing temperature and, consequently, allows its thermal management. This solution enables the use of the microcompressor technology under very high ambient temperatures as long as the condensing temperature is kept below 80°C (microcompressor shell temperature limit). Furthermore, the solution also allows the microcompressor to be confined, reducing the cooling unit noise level. The photograph of the cooling unit with its internal parts can be seen in Figure 4.

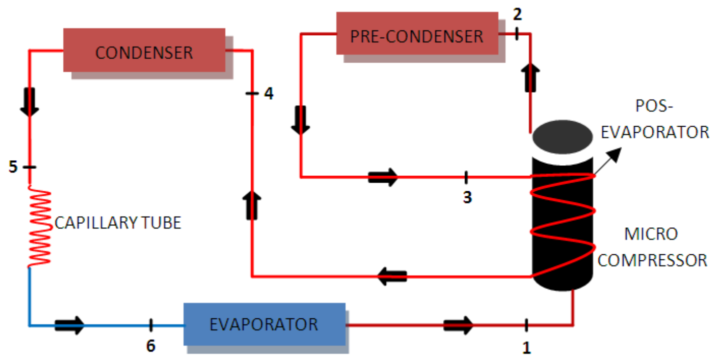

Figure 2. Squematic representation of the refrigerating loop.

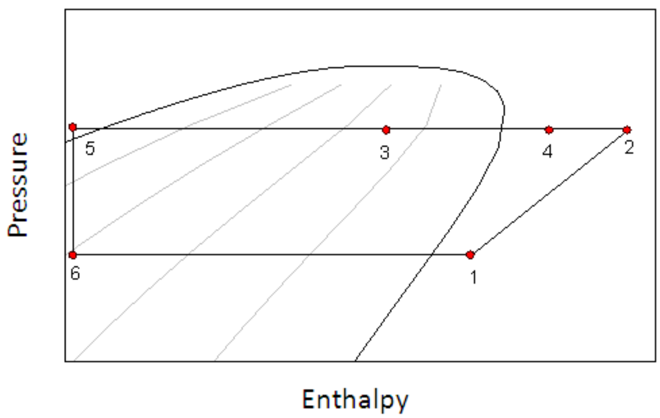

Figure 3. Pressure-enthalpy diagram of the refrigerating loop;

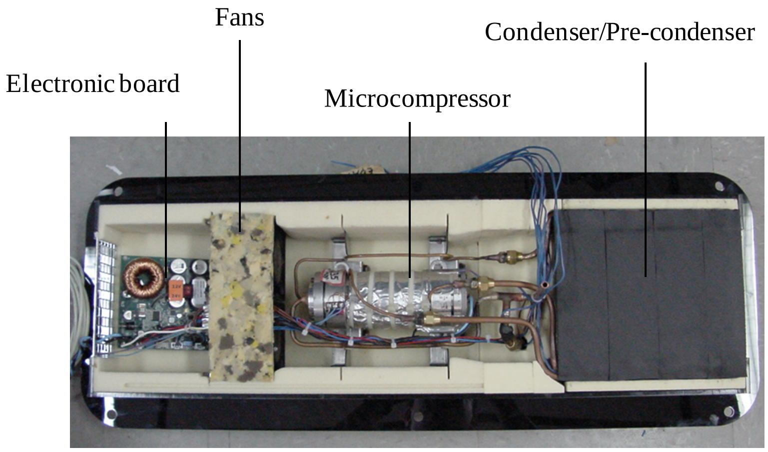

Figure 4. Microcompressor cooling unit in details.

2.2 Microcompressor

The microcompressor used in the unit is manufactured by Embraco and consists of an electronically-controlled hermetic linear compressor. It serves MBP (Medium Back Pressure), HBP (High Back Pressure) and VHBP (Very-High Back Pressure) applications, such as evaporating temperatures higher than -10 °C. The compressor is controlled by an electronic board (15 x 50 x 70 mm). The linear motion of the motor is transmitted to the piston by means of a resonant spring.

One of the compressor´s main features is orientation independence, which means it can operate in any position, regardless the gravitational orientation. This is possible because the compressor does not require any fluid as lubricant. This feature allowed the compressor to be installed in a vertical orientation, reducing the overall width of the unit. As can be seen in Figure 5, it has a cylindrical form (diameter of 60 mm and 160 mm in length) and weights 1.3 kg.

Figure 5. Linear microcompressor.

2.3 Heat Exchangers

The evaporator and the set condenser/pre-condenser used in this study were finned-tube heat exchangers with louvered fins as air-side heat transfer enhancement technique (Webb, 2005). The photograph of the heat exchangers is exhibited in Figure 6.

The condenser has a tubing circuit of six transversal tubes and two longitudinal tubes, whereas the pre-condenser has two tubes transversal and two tubes longitudinal to the air flow. The evaporator has a tube array of 5 x 2 (transversal x longitudinal). The tube pitch and the depth row pitch are 25 mm and 22 mm, respectively and both evaporator and condenser tube banks were arranged in-line. Standard copper tubes of 5/16” were used to produce the heat exchangers.

Moreover, the louvered fins present a fin thickness of 0.1 mm, pitch of 2 mm and were made of aluminum. The air flow through each heat exchanger was supplied by 24 Vdc compact fans manufactured by EBMPAPST. Two fans model 8414N were used for the evaporator whereas two fans model 8214 JN were applied for the condenser/pre-condenser set. Air flow rates of 30 m^3/h and 51 m^3/h were achieved for the evaporator and condenser/pre-condenser, respectively.

2.4 Electronic Control Unit

The electronic board consists in a printed circuit board that has an input voltage that can be varied between 9 to 36 Vdc. Beside the piston displacement control, the electronic board enables the temperature measurement via thermoresistors placed at the evaporator inlet and at compressor shell. The board also comes with a power supply of 24 Vdc/13 W in which the heat exchangers fans are powered and a communication between the cooling unit and the computer is possible through a RS-232 serial cable attached to the board.

For this latter purpose, a graphical interface based on the Labview software was created to control the cooling unit and is shown in Figure 7. With the interface, an on/off routine can be applied to control the enclosure ambient where electronics were installed.

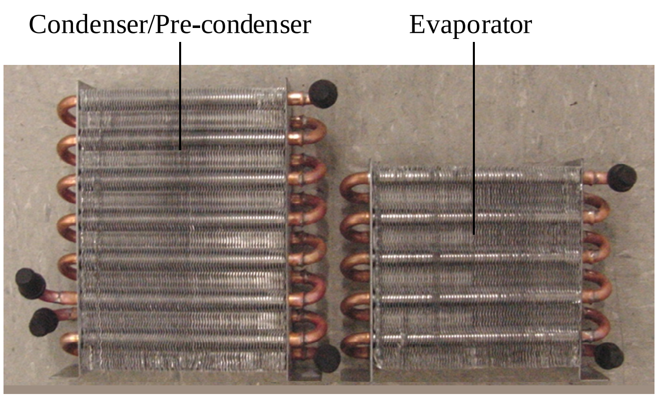

Figure 6. Photograph of the finned-tube heat exchangers.

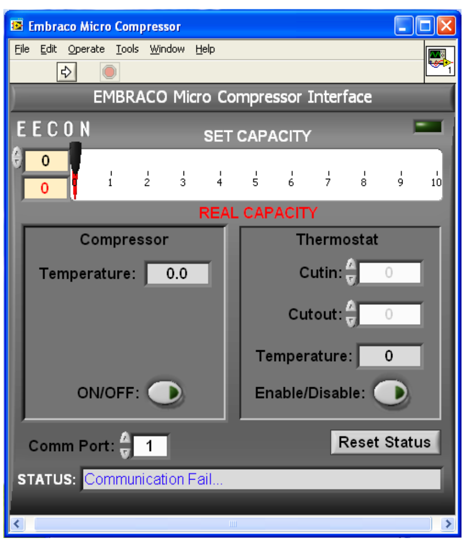

Figure 7. Cooling unit graphic interface.

3. EXPERIMENTAL APPARATUS

The cooling unit was evaluated using a calorimeter that emulates small enclosures found in electronics cooling. It consists in a cabinet insulated with polyurethane foam with an electric heater and an auxiliary fan inside. Twelve type-T thermocouples were installed inside the calorimeter and an electric heater was placed at the evaporator outlet. The cooling unit was attached to the calorimeter and placed inside an environmental chamber where the temperature and humidity are kept constant. The temperature inside the calorimeter is controlled manually via the heater. A power transducer measures its total consumption power.

Moreover, a variable-speed fan was placed inside the calorimeter as a manner to control the air pressure drop between the evaporator inlet and outlet, which is measured by a differential pressure transducer. The calorimeter and its devices and measurement instruments are shown schematically in Figure 8.

A total of seven type-T thermocouples were installed along the refrigeration circuit. Two thermocouples were placed at evaporator inlet and outlet. Furthermore, three thermocouples were placed at condenser inlet, outlet and middle, while the remaining thermocouples were fixed at the compressor suction line and shell. To complete the full instrumentation of the refrigeration system, a power transducer measured the power input consumed by the cooling unit.

The cooling unit is instrumented and connected at the bottom of the calorimeter. Finally, the calorimeter is placed inside the environmental chamber.

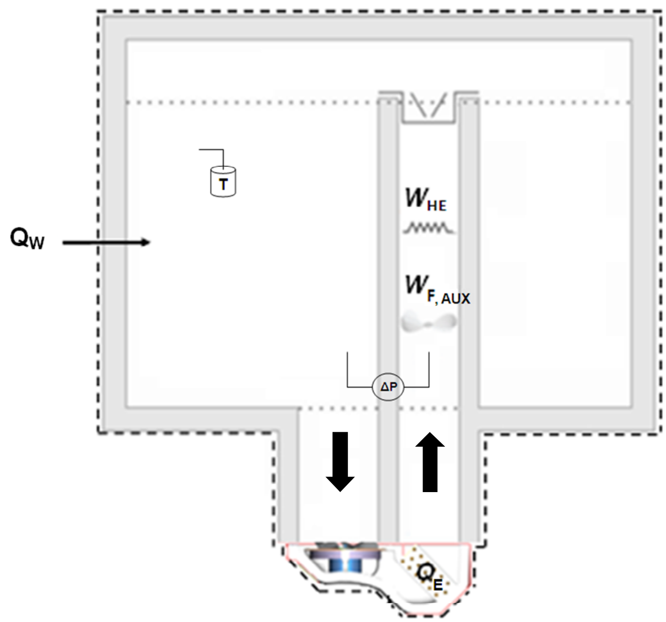

Figure 8. Schematic representation of the calorimeter.

4. EXPERIMENTAL PROCEDURE

For the test runs of the microcompressor cooling unit, the ambient temperature controlled by the environmental chamber was varied at three temperatures: 25, 35 and 55 °C. On the other hand, temperatures of 25 and 35 °C were applied for thermoelectric units evaluation. For all cases, the enclosure temperature was maintained at the same temperature of the ambient by means of the electric heater placed inside the calorimeter, while the air pressure drop between evaporator inlet and outlet was kept close to zero varying the speed of the auxiliary fan. The microcompressor unit was evaluated under full stroke (i.e, maximum piston displacement), and the refrigerant mass and the determination of the length of the capillary tube were specified at the ambient temperature of 35 °C.

Overall, three evaluation tests were performed for the microcompressor cooling unit and seven tests were used at the analysis. The cooling capacity and the coefficient of performance under different ambient/enclosure temperatures are the performance factors used in this study.

5. DATA REDUCTION

As described by Gosney (1982), for thermodynamic cycles the coefficient of performance (COP) is defined as the ratio of the desired heat output to the required work input. At this particular case the heat output is considered as heat absorbed by the evaporator, in other words, cooling capacity. The work input is represented as power input consumed by the cooling unit. COP is commonly used as figure of merit for the assessment of refrigerators and air-conditioners in general. Additionally, the COP is calculated as:

$$COP = \frac{\dot{Q}_E}{W_{CU}} \tag{1}$$

where $\dot{Q}_E$ is the cooling capacity and W_{CU} is the cooling unit power input. The latter is measured by the power transducer and the former is obtained by a simple energy balance indicated in Figure 8 as follows:

$$\dot{Q}_E = \dot{Q}_W + W_{HE} + W_{F,AUX} \tag{2}$$

where W_{HE} and $W_{F,AUX}$ are the heat dissipated by the electric heater and by the auxiliary fan, respectively. The term $\dot{Q}_W$ represents the heat transfer rate between the calorimeter and surroundings. Even though all duct walls were carefully insulated with polyurethane foam the large surface areas resulted in heat exchanges that could not be avoided.

In order to estimate the heat losses/gains across calorimeter walls, a test without the use of a cooling unit was carried out. By means of this particular test, the global thermal conductance associated with the calorimeter walls UA_{CAL} was obtained e defined using eq. (3) and (4) as

$$\dot{Q}_W = W_{HE} + W_{F,AUX} \tag{3}$$

$$UA_{CAL} = \frac{\dot{Q}_W}{(T_{ENC} - T_{AMB})} \tag{4}$$

where T_{ENC} and T_{AMB} are the enclosure and the ambient temperatures, respectively. For the UA_{CAL} determination, the ambient temperature was kept in 25 °C, 60 W was dissipated inside the enclosure and the average enclosure temperature was stabilized in 51 °C. Thus, a global thermal conductance of 2.3 W/°C was found and the heat transfer rate $\dot{Q}_W$ is calculated as

$$\dot{Q}_W = 2.3(T_{AMB} - T_{ENC}) \tag{5}$$

6. RESULTS

The test used to specify the capillary tube and the refrigerant mass was performed, yielding a capillary tube of 0.518 mm of diameter and 1.15 m of length. The optimized refrigerant charge was 138 g.

Table 1 presents the results achieved with the thermoelectric cooling units and microcompressor unit using full stroke, at the ambient/enclosure condition of 35 °C. As can be seen, the thermoelectric units 1 and 2 reached cooling capacities of 60.5 and 145 W, respectively. The test performed with the microcompressor at the full stroke presented an intermediate cooling capacity of 120.4 W. However, the vapor compression unit presented the COP around 2 times higher than thermoelectric solutions. Furthermore, close values of the compressor shell temperature and condensing temperature (represented by the condenser centre temperature) indicate that the proposed refrigerating loop acts like a thermal management mechanism.

Table 1. Results for the ambient/enclosure temperature of 35°C.

Cooling unit	Thermoelectric unit 1	Thermoelectric unit 2	Microcompressor unit
Ambient [°C]	35.1	35.1	34.9
Enclosure [°C]	35.7	35.0	35.1
Condenser inlet [°C]	-	-	59.3
Condenser middle [°C]	-	-	50.4
Condenser outlet [°C]	-	-	48.8
Evaporator inlet [°C]	-	-	18.8
Evaporator outlet [°C]	-	-	16.4
Suction line [°C]	-	-	28.3
Shell temperature [°C]	-	-	59.1
Calorimeter power input [W]	61.9	149.8	120.9
Heat losses [W]	-1.4	0.2	-0.5
Cooling capacity [W]	60.5	150.0	120.4
System power input [W]	104.3	283.0	110.4
COP [-]	0.58	0.53	1.09

Figure 9 presents the behavior of the cooling capacity as function of the ambient/enclosure temperature. For all refrigerating system configurations, the cooling capacity slightly increases with the increasing ambient temperature. According to Gosney (1982), the cooling capacity of a general refrigerating system tends to increase with the rise of evaporating temperature, as well as the decreasing condensing temperature. Since evaporating and condensing temperatures increase with the environmental temperatures (i.e, ambient and enclosure temperature), the net effect resulted in a higher influence of the evaporating temperature for all studied cooling units.

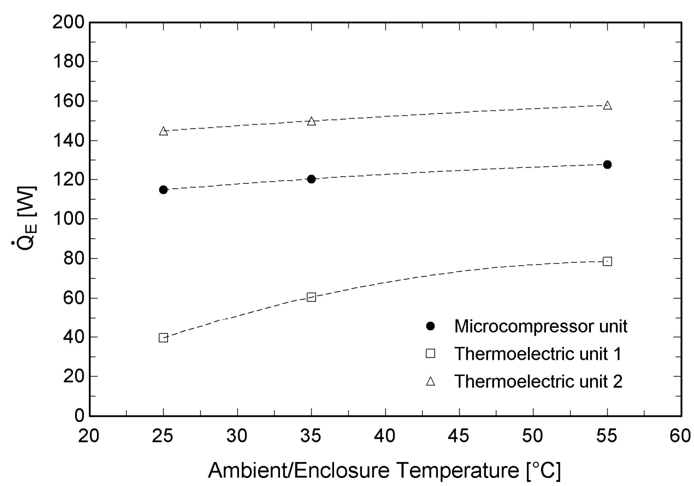

Figure 9. Cooling capacity of cooling units.

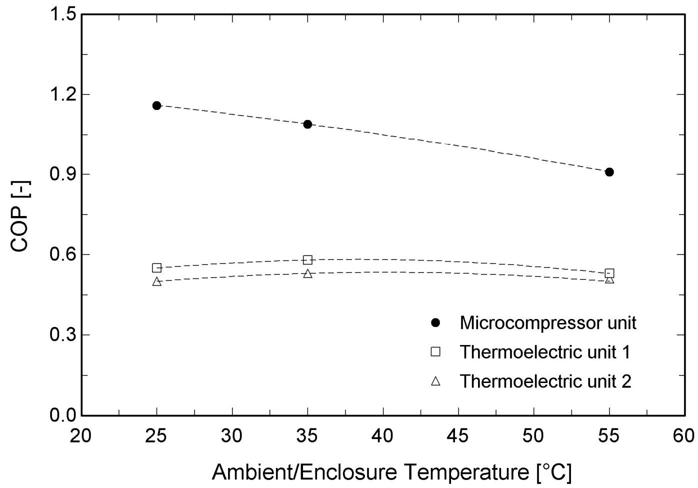

Figure 10. Coefficient of performance of cooling units.

Results for the COP are shown in Figure 10. As described before, microcompressor unit working at full stroke presented the highest COP, for all environmental conditions. Unlike the cooling capacity, the COP of the microcompressor cooling unit decreased with the increasing ambient/enclosure temperatures. These results can be justified by the fact that the linear motor efficiency tends to diminish for high shaft powers.

Considering that all measurements are random and uncorrelated, the uncertainty propagation analysis with a confidence level of 95% was performed based on the data reduction equations. The absolute uncertainty for differential pressure transducer was 2.49 Pa, while the uncertainties for the thermocouples and for the power transducer were 0.50 °C and 1.01 W. Finally, the estimated measurement uncertainties of the cooling capacity and COP were ±1.75 % and ±1.98%, respectively.

7. CONCLUSIONS

In order to consider the feasibility of using the microcompressor in a refrigerating system for electronics cooling, tests were performed and results were compared against commercially available thermoelectric systems.

The microcompressor uses a linear motor that controlled electronically by a printed circuit board. Finned-tube heat exchangers and the capillary tube are the remaining components used in the refrigeration circuit. The system operated with R-134a and a singular refrigerating loop was created in order to manage the compressor shell temperature for extreme conditions (i.e, high ambient temperatures).

A cooling unit calorimeter was designed for evaluation purposes and the cooling capacity and COP were calculated by means of measurements results. Ambient temperatures ranged from 25 °C to 55 °C. It was observed that the cooling capacity increased with the increasing ambient and enclosure temperatures. On the opposite, higher COP was achieved when the microcompressor worked at a lower ambient/enclosure temperature. Moreover, results show that the microcompressor presented COP approximately two times higher than the thermoelectric solution applied nowadays.

NOMENCLATURE

COP	coefficient of performance	(–)
$\dot{Q}$	heat transfer rate	(W)
T	temperature	(°C)
UA	thermal conductance	(W/°C)
W	power input	(W)

Subscripts

AMB	ambient
AUX	auxiliary
CAL	calorimeter
CU	cooling unit
E	evaporator
ENC	enclosure
F	fan
HE	heater
W	walls

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